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# MODELING, MEASUREMENTS AND ANALYSIS OF GAS-FLOW GENERATED NOISE FROM TWIN-SCREW COMPRESSORS

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## ABSTRACT

This paper gives a short presentation of a method for the computer simulation of noise caused by the gas pulsations, created from the opening and closing of inlet and discharge ports of twin-screw compressors. The method takes into account influences from lobe combinations, wrap angle, length/diameter ratio, leakage through blowhole and clearances at mesh, rotor tips and discharge end. Oil content in the gas as well as resonance in the compressor discharge chamber and downstream piping are also taken into consideration. Comparisons between calculations and laboratory measurements, as well as analysis of the influence of different design parameters, are presented and discussed.

## NOMENCLATURE

$p(t)$	pressure pulsation amplitude in inlet or discharge pipe	$S$	cross sectional area of pipe
$\rho$	density of gas	$Q(t)$	volume flow in pipe
$c$	velocity of sound	$\bar{Q}$	average volume flow in pipe

## INTRODUCTION

The noise generated from the operation of twin screw compressors is a crucial parameter for the design of systems, where the need for quiet operation is an important factor. The noise level determines the size and form of such equipment as silencers, pipes etc., that are included in the system. Especially in the case of new applications, it is essential to predict and analyze the noise and vibration problems that may occur.

Since the main source for the generation of noise is the gas pulsations caused by the opening and closing of inlet and discharge ports, it is important to include the time-dependent area variation of these ports in the calculations as well as the influence of other design parameters such as number of lobes, wrap angle, length/diameter ratio, built-in volume ratio, leakage through blowhole, mesh, rotor tips, end clearances, volume variation, oil content in the gas, discharge chamber design and upstream/downstream piping.

This paper therefore gives a short presentation of a simple model which is an extension of the theory as described in [1]. The complete model is a useful tool to get a better understanding of the mechanism which governs the behaviour of the pressure pulses in the inlet or discharge piping connected to a particular machine. For more detailed information see ref. [2], [3] and [4].

## THEORY

This time-dependent mass flow and the pressure oscillations in the connecting pipe can be calculated with the model according to fig.1.

In this model it is assumed that the discharge pipe is of infinite length, which means that no wave reflections can occur in it, but the possibility for resonance in the chamber within the housing still exists.

The pressure oscillations in the connecting pipe can then be calculated with the following equation:

$$p(t) = \frac{\rho \cdot c}{S} (Q(t) - \bar{Q})$$

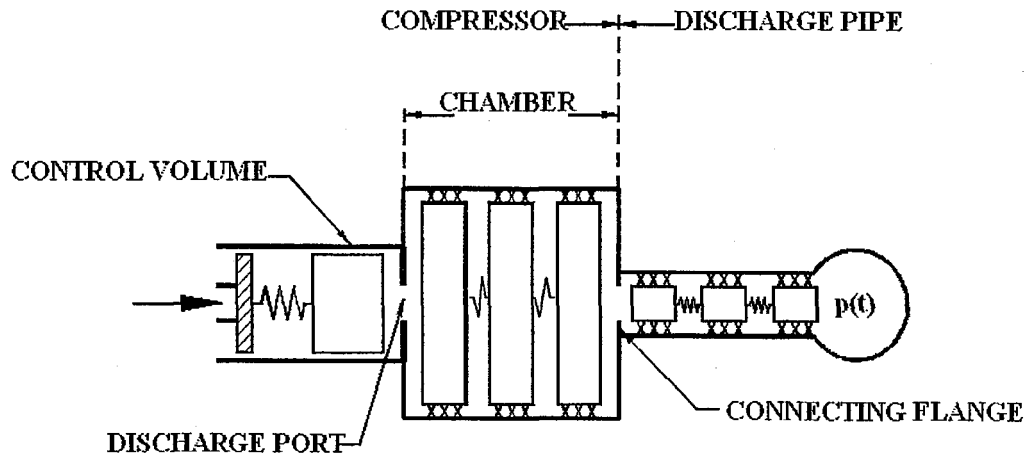


Fig. 1. Discharge model.

Coupling of the differential equations for this model, described in [2] and [3] to the differential equations described in [1] gives a new model for calculating the influence of the twin-screw compressor design parameters on the pressure oscillations in the discharge or inlet pipes.

## COMPARISON OF CALCULATION RESULTS WITH LABORATORY TEST DATA FOR A DRY SCREW COMPRESSOR.

The screw compressor previously tested is equipped with long connecting pipes containing absorbers. The arrangement prevents wave reflections, which is in accordance with the theoretical assumptions for the above mentioned model.

Table 1 shows measured and calculated sound pressure levels in the inlet and discharge pipes of a 315 mm 3+4 unequal bore dry screw compressor operating at 3000 rpm. Inlet/outlet pressure is 1.0/1.4.

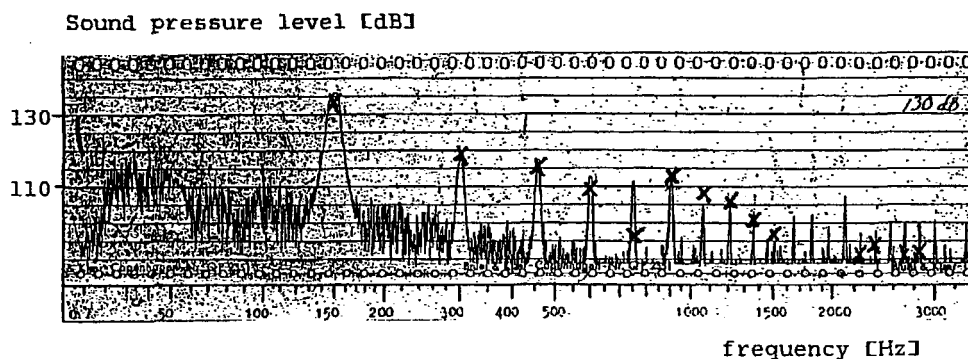
Table 1. Sound pressure levels from a 3+4 screw compressor.

	Sound Pressure Level (dB)	
	measured	calculated
Outlet	164.0	163.9
Inlet	139.0	139.3

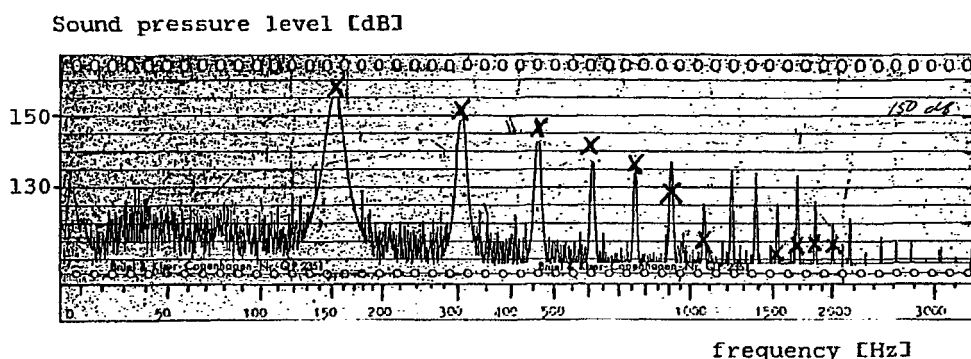
Fig.2 presents recordings from a narrow-band-analysis of the measurements of this screw compressor compared to the Fourier analysis of the computed values.

It is evident that measurements and calculations are very close for the fundamental and the first three harmonics.

### INLET PIPE



### DISCHARGE PIPE



X = Fourier analysis of calculated screw compressor sound.

Fig. 2. Narrow-band analysis of screw compressor sound.

### ANALYSIS OF A DRY COMPRESSOR

Table 2 shows measured and calculated sound pressure levels in the inlet and discharge pipes of a 200 mm 4+6 equal bore dry screw compressor operating at 7500 rpm.

Table 2. Sound pressure levels from a 4+6 screw compressor.

Outlet Pressure (bar)	Outlet Sound Pressure Level (dB)	
	Measured	calculated
1	165.0	165.4
3	160.0	159.9
Inlet Pressure (bar)	Inlet Sound Pressure Level (dB)	
	Measured	calculated
1	142.0	141.1

In order to demonstrate the possibilities of the computer method, this dry screw compressor is analysed regarding the effects of different design parameters on the sound pressure level. In the analysis, the following parameters have been investigated:

- Leakage
- Wrap angle
- Length/Diameter ratio
- Lobe combination

The investigations are presented so as to show the effect of a parameter change. All simulations are made with constant compressor swept volume and rotor speed. This is possible by compensating the effect of the actual parameter change by a change of rotor diameter. In all investigations the clearances are held constant, except for the case "Influence of leakage".

#### **Influence of Leakage**

Table 3 shows the influence on sound pressure level in the discharge pipe when the flow through the different types of leakage paths is changed from normal to zero. As a result of the increased flow through the screw compressor the sound pressure level is raised.

The noise level for the basic design = 159.9 dB.

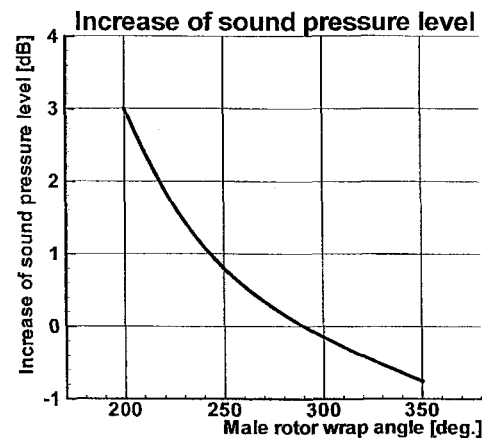
**Table 3. Influence of leakage on sound pressure level at discharge.**

INCREASE OF SOUND PRESSURE LEVEL (dB)	
Without meshing leakage	1.6
Without end plane leakage	0.2
Without lobe tip leakage	1.2
Without blow hole leakage	0.2

Elimination of these leakages increases the inlet noise by approximately 1 dB.

#### **Influence of Wrap Angle**

Fig.3. presents how the wrap angle affects the sound pressure level at the compressor discharge. Increased wrap angle reduces the sound pressure level. The explanation for this is that an increase of wrap angle prolongs the opening time for the discharge port, which leads to a softer discharge. 150 deg. increase of wrap angle decreases the inlet noise level by approximately 1 dB.



**Fig. 3. Influence of wrap angle on sound pressure level at discharge.**

#### **Influence of Length/Diameter ratio**

The influence of this parameter on the discharge and inlet noise is less than 0.1 dB, i.e. within the limits of accuracy of practical measurements. The effect is expected, because a change of L/D does not change the opening time for the discharge and inlet ports.

### Influence of Lobe Combination

Table 4 shows the effect of different lobe combinations on the inlet and discharge pressure levels. As can be seen, the combinations with many lobes - giving more continuous filling and discharge - generate the least noise.

**Table 4. Influence of lobe combination.**

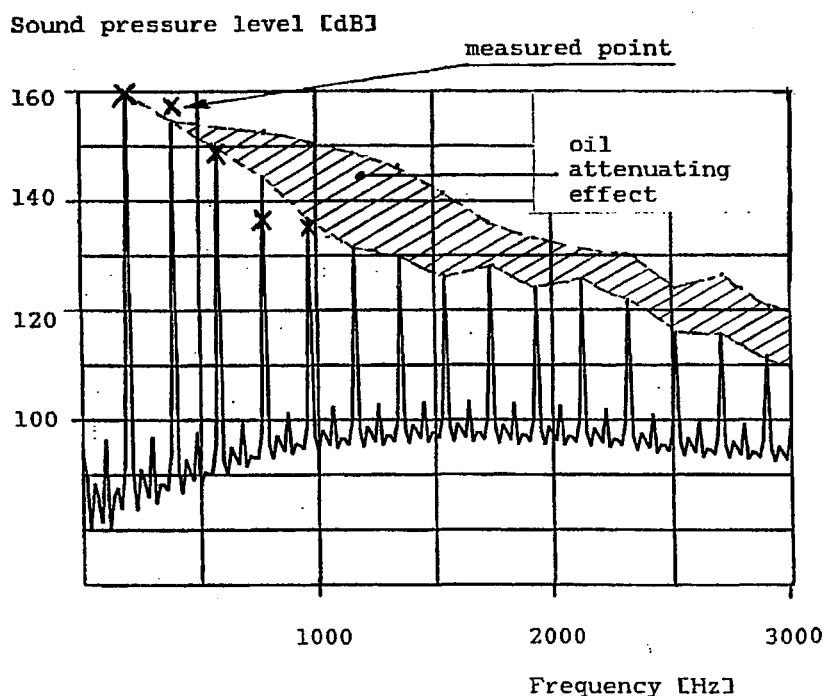
Lobe combination	Sound pressure level (dB)	
	Inlet	Outlet
3 + 4	148.0	161.4
3 + 5	148.7	161.3
4 + 6	142.9	159.9
5 + 7	140.2	158.7
5 + 6	140.7	158.6
6 + 8	141.1	158.4
6 + 7	141.1	158.7

### **ANALYSIS OF AN OIL-INJECTED SCREW COMPRESSOR**

Most compressors on the market are oil-injected. It is therefore of interest to investigate how the amount of injected oil influences the sound pressure level.

Fig.4 shows this effect on the Fourier analysis of the discharge sound from a 4+6 equal bore 204 mm oil injected screw compressor for air operating at 2900 rpm. The fundamental and the first harmonic are not influenced by the oil-injection, but the rest of the harmonics are attenuated. In fig.4 results from practical measurements are also plotted.

As can be seen, the calculated values are very close to those measured.



**Fig. 4. Fourier analysis of discharge sound from an oil-injected screw compressor.**

## COMPARISON OF CALCULATED RESULTS WITH LABORATORY TEST DATA FOR AN OIL-INJECTED SCREW COMPRESSOR WITH RESONANCE IN THE DISCHARGE CHAMBER.

Fig.5 shows the calculated and measured sound pressure levels at different discharge pressures for a 4+6 equal bore 245 mm screw compressor operating at 3000 rpm. It should be pointed out that the predicted minimum value is not at 8 bar even though the inlet pressure is 1 bar and the built-in pressure ratio is 8:1.

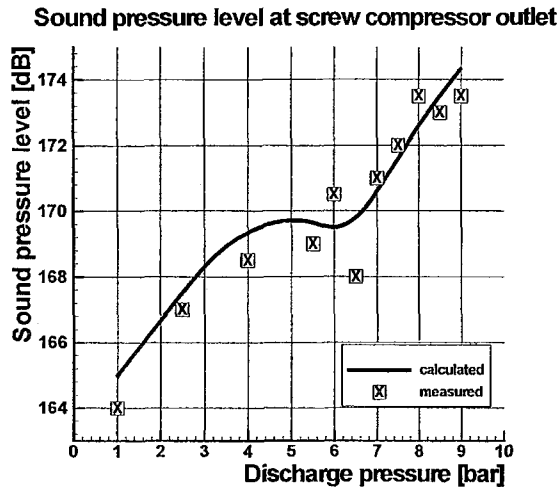


Fig. 5. Influence of discharge pressure on sound pressure level in the discharge pipe.

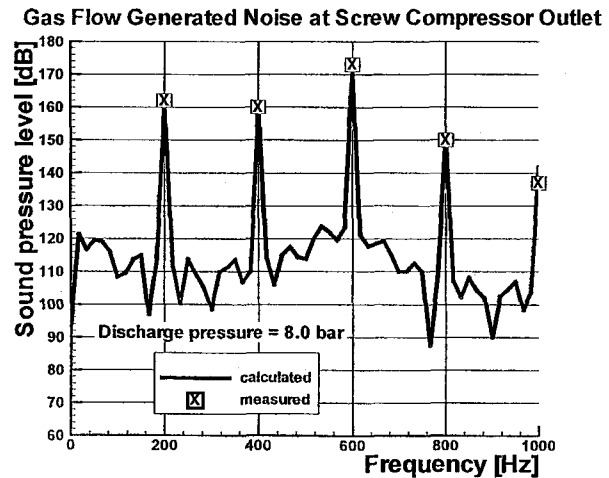


Fig. 6. Fourier analysis of the sound pressure level in the discharge pipe.

A comparison between the frequency analysis of the calculated and the measured noise at discharge pressure 8 bar shows good correlation, as is evident from fig.6.

## ANALYSIS

In order to find out whether there is any possibility of reducing the noise level in the discharge pipe of this screw compressor by modified design, calculations were made for different discharge pressures and pipe lengths of the chamber. See fig.7.

According to these calculations, at discharge pressures above 6 bar, a chamber pipe length of 0.2 m gives the lowest sound pressure level, while 0.3 m gives the highest.

It may be seen that in all cases, despite the built-in discharge pressure being 8 bar for an inlet pressure of 1 bar, the minimum occurs at a discharge pressure of 6 bar. This corresponds to the results already shown in fig.5 and fig.7.

It can also be seen that for the chamber length of 0.3 m, the curve has its lowest value at low discharge pressure.

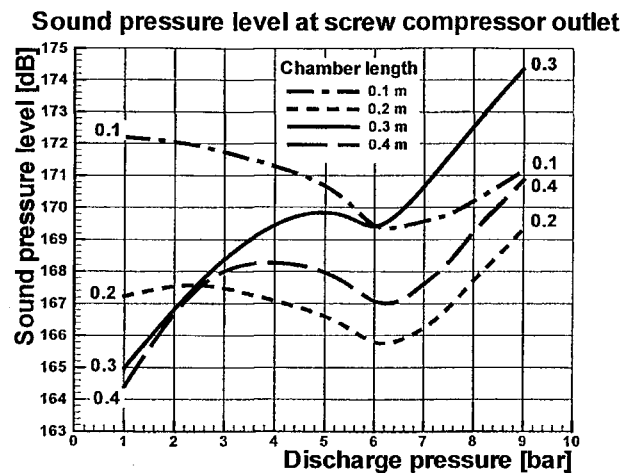


Fig. 7. Influence of discharge chamber length at different discharge pressures.

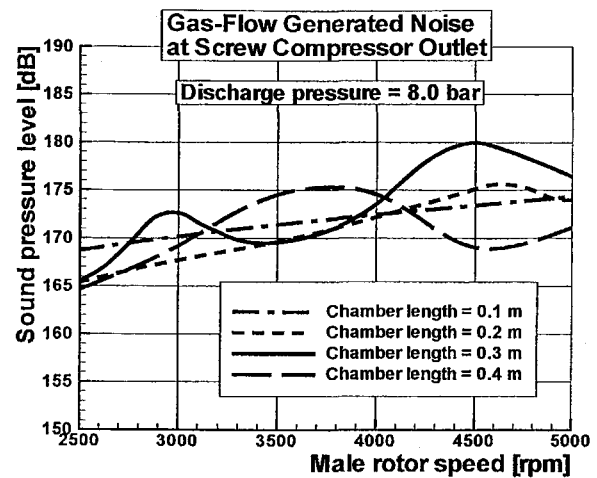


Fig. 8. Influence of discharge chamber length at different male rotor speed.

The explanation for this seems to be that the resonance has broken down at these pressures, while it exists very strongly at high discharge pressures.

Fig.8 shows that a chamber length of 0.2 m gives the lowest sound pressure level even if the male rotor speed varies from 2500 to 4000 rpm. In this speed range there are no resonance peaks for 0.2 m, but for 0.3 and 0.4 m there are one.

## CONCLUSIONS

A computer method for the calculation of gas flow generated noise has been presented.

Comparison between calculations and laboratory tests show good correlation.

An analysis of a 200 mm dry screw compressor with regard to the influence from different design parameters has been presented and shows:

- A screw compressor designed with rotors with many lobes has generally a lower sound pressure level in operation than a screw compressor with few lobes. However, many lobes means high frequency, which is harder to listen to than a low one.
- Decreased leakage results in increased noise generation at the discharge, due to increased flow through the screw compressor, but the change is within the limits of accuracy of practical measurement.
- A change of wrap angle of 50 degrees will change the discharge noise level by approximately 1 dB, which is within the accuracy-limits of practical measurements.
- A change of Length/Diameter ratio has no practical influence on generated noise level.

An analysis of the influence of the injected oil in a screw compressor for air, shows that the sound pressure level of the fundamental is not influenced by the oil, but the harmonics are attenuated.

An analysis of a 245 mm diameter oil-injected screw compressor tested at SRM, shows that the chamber between the outlet port and the connected discharge pipe creates resonance at 3000 rpm. The analysis and the measurements show that the lowest sound pressure level is not obtained at the optimum built-in pressure ratio, which was expected. The analysis also shows that a proper design of the chamber between the outlet port and the connected pipe can reduce the sound pressure level in the discharge pipe by 5 dB.



## REFERENCES

1. B. Sångfors  
"Computer Simulation of the Oil Injected Twin Screw Compressor"  
Proceedings of the 1984 Purdue Compressor Technology Conference, pp 528 - 535.
2. D.L.Margolis  
"An Analytical Study of Intake Tuning for Helical-Rotor Expanders"  
Lawrence Livermore Laboratory, Rpt. UCRL-52449 1978.
3. D.L.Margolis  
"Modeling of Two-Stroke Internal Combustion Engine Dynamics  
using the Bond-Graph Technique"  
Transactions of SAE, pp 2263-2275 September 1975.
4. B. Sångfors  
"Computer Simulation of Gas-Flow Generated Noise from Twin-Screw Compressors"  
Proceedings of the 1999 ImechE Int .Conf on Compressors and their Systems, pp 707 -716.